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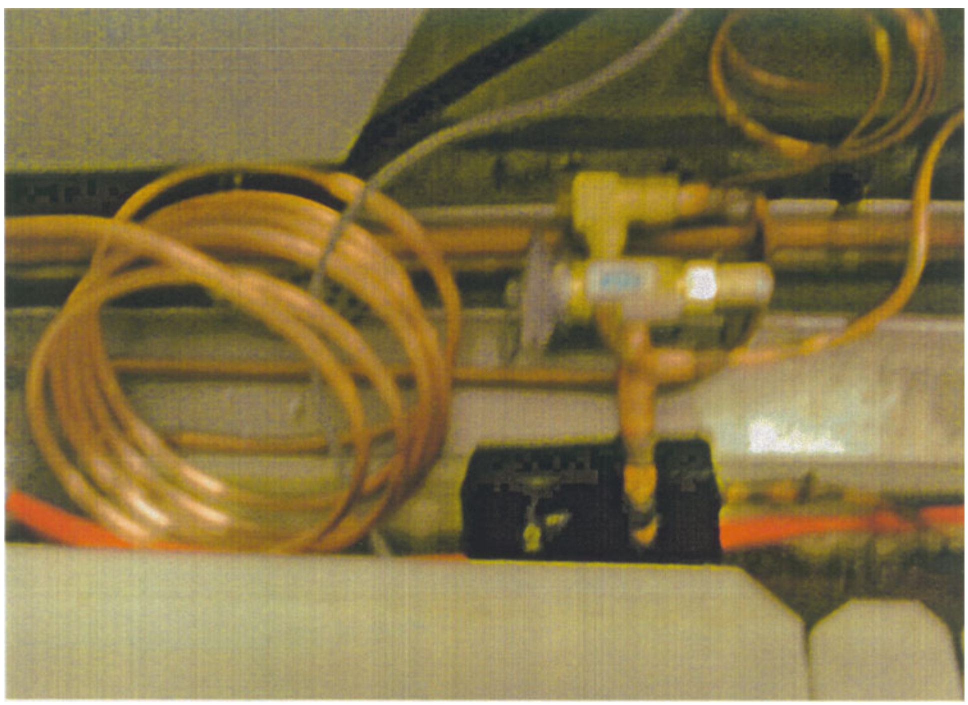
Evaluation of a Multifunctional Valve Assembly in a Direct Expansion Refrigeration System



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Evaluation of a Multifunctional Valve Assembly in a Direct Expansion Refrigeration System

Phase 1&2 Report

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Abstract

A multifunctional valve (MXV) assembly [consisting of additional liquid line, an XTC valve, and a larger thermostatic expansion valve (TXV)] was installed on all display cases of an instrumented supermarket refrigeration test rig. The refrigeration test rig includes two low-temperature single-deck display refrigerators; two 2-door reach-in cases; and a condensing unit with three unequal compressors, a water-cooled condenser, a water-cooled subcooler, an oil management system, and a programmable controller. Tests were performed at various combinations of evaporating temperature (-30 or -27 °F), condensing temperature (75 or 105 °F), and defrost schedules (once per 24 or 48 hours) under either temperature or pressure control. Results were compared to tests at the same conditions on the baseline system.

Lower package temperatures were achieved under pressure control with the MVX assembly due to the lower superheats specified by the MXV manufacturer, but this reduction came at an energy penalty. Under temperature control—the control methodology used in field applications—there was no energy or product temperature benefit seen with the MXV valve assembly. Although coil pull-down times after defrost were shorter, there was no impact on daily energy use. Both the MXV and baseline system performed well with one defrost per 48 hours, and each had about 4% energy savings compared to a more frequent defrost schedule. However, at this condition, MXV showed no added benefit over baseline.

Foreword

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Lawrence W. Reiter, Acting Director
National Risk Management Research Laboratory

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Executive Summary

Introduction

New expansion devices have the potential to improve the performance and possibly to reduce energy usage of supermarket display cases. One suggested device is a multifunctional valve (MXV) assembly[1]. Tests of this valve assembly were requested to investigate its performance and impact on energy consumption and package temperatures compared to the performance of conventional thermostatic expansion valves that are presently used in supermarket display cases.

Test Equipment and Procedure

The MXV assembly [consisting of an additional liquid line, an XTC valve, and a thermostatic expansion valve (TXV)[2]] was installed on all display cases of an instrumented supermarket refrigeration test rig. The refrigeration test rig includes two low-temperature single-deck display refrigerators; two two-door reach-in cases; and a condensing unit with three unequal compressors, a water-cooled condenser, a water-cooled subcooler, an oil management system, and a programmable controller. Tests were performed at various combinations of evaporating temperature (-30 °F or -27 °F), condensing temperature (75 °F or 105 °F), and defrost schedules (once per 24 or 48 hours) under either temperature or pressure control. For the valve assembly tests, superheats were set to 0–5 R (per manufacturer's instructions) while for the baseline tests superheats were 8–10 R. The primary evaluation criteria were the product temperatures and the energy consumption.

The MXV assembly was tested at the same conditions as earlier baseline tests and then at additional conditions which the manufacturer felt would show the MXV benefits. These additional conditions included replacement of the existing temperature controller, testing with one defrost per 48 hours, and testing under higher suction pressure. These new conditions required a few additional baseline tests to be performed after the MXV tests were completed.

Superheats for the MXV assembly tests could be properly adjusted only in pressure control when case solenoid valves remain fully open.

Results and Discussion

MXV assembly tests were performed and compared to the baseline system. The following prefix designations are used in Table 1: BS for baseline system tests, X for tests with the full MXV valve assembly, PC for tests under pressure control, and TC for tests under temperature control.

Table 1. Comparison of MXV Assembly Tests and Baseline System Tests

Control	Evap. °F	Cond. °F	Defrost Schedule	Test	Energy kWh/day	Package Temperatures (°F)			
						Case 1	Case 2	Case 3	Case 4
Press.	-30	75	1/24 h	BSPC1	97.2	-14.1	-12.8	-6.7	-8.3
				XPC1	101.2	-15.5	-14.9	-8.2	-9.9
Temp	-30	75	1/24 h	BSTC1	97.9	-10.2	-10.1	-5.9	-7.1
				XTC1	97.0	-10.2	-9.7	-6.3	-6.7
				XTC2	98.9	-10.4	-10.2	-6.0	-6.4
Temp	-30	75	1/48 h	$BSTC3_{avg}$	94.0	-10.4	-10.2	-7.4	-7.7
				$XTC3_{avg}$	93.5	-10.2	-9.5	-7.7	-7.8
Temp	-27	75	1/24 h	BSTC4	94.3	-10.4	-10.1	-5.1	-6.3
				XTC4	94.3	-9.8	-9.8	-5.1	-5.6
Temp	-27	105	1/24 h	BSTC5	125.8	-10.5	-10.4	-4.4	-6.7
				XTC5	125.5	-10.2	-10.0	-4.6	-5.8

Pressure Control Tests

The MXV assembly (XPC1) uses 3% more energy than the baseline system (BSPC1); however, package temperatures are about 1.5 °F lower. These lower package temperatures are a result of the lower superheats specified by the manufacturer in the MXV tests.

Temperature Control Tests

Energy consumption for both the MXV and baseline tests are comparable under the original temperature control strategy (XTC1 and BSTC1). MXV package temperatures are also comparable to baseline system tests. Under the alternate temperature controller (XTC2), MXV energy consumption is slightly higher than with the original controller.

Pull-down times after defrost (defined by the coil reaching set-point to start solenoid valve cycling) were about 50% shorter in the MXV tests than in the baseline tests. By comparing

XTC and BSTC, tests it can be seen that this faster pull-down time for the coil did not result in energy savings.

Reducing the defrost schedule to once per 48 hours reduced the energy consumption of both systems by about 4% compared to one defrost per 24 hour. Energy use and package temperatures of the MXV and baseline systems were comparable under a 48 hour defrost schedule.

Higher Suction Pressure

The MXV manufacturer suggested that, since the MXV tests under pressure control showed lower product temperature than the baseline system, the suction pressure could be raised in the temperature control tests with no degradation of product temperature but with some energy savings. Suction pressure was raised by 2 psi to give a nominal evaporating temperature of -27 °F. At this condition, energy consumption and package temperatures were comparable between the MXV and baseline systems.

Higher Condensing Temperature

At 105 °F condensing, energy consumption for both the baseline and the MXV valve assembly systems are 30% higher than at 75 °F condensing. Energy use for the two systems is comparable with lower package temperatures for Case 4 in the baseline system.

Conclusion

Lower package temperatures were achieved under pressure control with the MXV assembly due to the lower superheats specified by the MXV manufacturer. This reduction came at an energy penalty.

Under temperature control—the control methodology used in field applications—there was no energy or product temperature benefit seen with the MXV valve assembly. Although coil pull-down times after defrost were shorter, there was no impact on daily energy use.

Both the MXV and baseline systems performed well with one defrost per 48 hours, and each had about 4% energy savings compared to a more frequent defrost schedule. However, at this condition, MXV showed no added benefit over baseline.

Evaluation of a Multifunctional Valve Assembly in a Direct Expansion Refrigeration System

Introduction

As part of the U.S. support to the International Energy Agency's Annex 26 on Advanced Supermarket Refrigeration/Heat Recovery Systems, the U.S. Department of Energy (DOE) is gathering information on a variety of systems used or of potential use in supermarkets for refrigeration. The ultimate goal of this project is to demonstrate advanced systems with reduced refrigerant charge and reduced energy consumption compared to systems that are presently used. When investigating methods to reduce overall energy use, it is important to evaluate component options which can be used to improve energy efficiency. Major contributors to the energy demand in refrigeration systems are the display cases, and thus there is interest in investigating energy saving features for this equipment.

For the display case, new expansion devices have the potential to improve performance and reduce energy consumption. One suggested device is the MXV assembly[1]. This assembly includes a proprietary valve (XTC), an expansion assembly, and a power head or controller assembly[2]. Tests of this valve assembly were requested to investigate its performance and impact on energy consumption and package temperatures compared to the performance of conventional thermostatic expansion valves that are presently used in supermarket display cases.

Background

The U.S. Environmental Protection Agency has developed a highly instrumented supermarket refrigeration system. The system has about 300 measured parameters, including temperatures (using RTDs and thermocouples), pressures, mass flow, power input, and energies. Baseline data with conventional thermostatic expansion valves have been collected on this system at -30 °F evaporating temperature and at condensing temperatures from 50 to 105 °F. Tests have been performed at various levels of liquid subcooling. This system was retrofitted with the MXV assembly to evaluate its performance in open and reach-in cases.

Test Equipment and Procedure

The EPA facilities include an instrumented supermarket refrigeration test rig, chambers for environment control around the cases, two synchronized data acquisition systems, and a large chiller for condensing temperature control. The refrigeration test rig includes two low-temperature single-deck display refrigerators; two 2-door reach-in cases; and a condensing unit with three unequal compressors, a water-cooled condenser, a water-cooled subcooler, an oil management system, and a programmable controller. The primary factors of interest are the product temperatures and the energy consumed by the various components, including the total energy, to achieve the desired product temperatures. A schematic of the test rig is shown in Figure 1. A list of nomenclature for measured parameters is shown in Appendix C.

In order to simulate operation in a store, the cases are located in an environmental chamber where ambient temperature and relative humidity (RH) are maintained at 75 °F and 55% (64 °F wet bulb), respectively. To monitor product temperatures in the cases, test and dummy packages as prescribed in ASHRAE 72-1983R[3] and ASHRAE 117-1992R[4] are used. Test packages, thus product temperature sensors, are located in all four cases. Each reach-in case has 36 test packages, and each open case has 12. Energy consumption of each case (defrost beaters, fans, anti-sweat heaters, and lights) are also monitored. Energy use of the individual compressors is collected, as well as the total energy use of the refrigeration system.

The test rig also includes pneumatic door openers for the reach-in cases. These were set to open the doors a full 90° for ten seconds. Each door was opened six times per hour with a two-minute delay between each door, and openings continued for eight hours. Door operations occurred under all test conditions including the days when the system was in transition between test conditions. Similarly, conditions in the environmental chambers were monitored and maintained at all times including transition periods.

Test conditions are specified by setting evaporating temperature (suction pressure at the compressor manifold), coil superheats, condensing temperature, and liquid subcooling. For all tests, liquid subcooling was set to 8 R below the condensing temperature. For the baseline system tests, superheats were set to 8–10 R at each case. For the MXV tests, the manufacturer requires superheats to be set between 0 and 5 R.

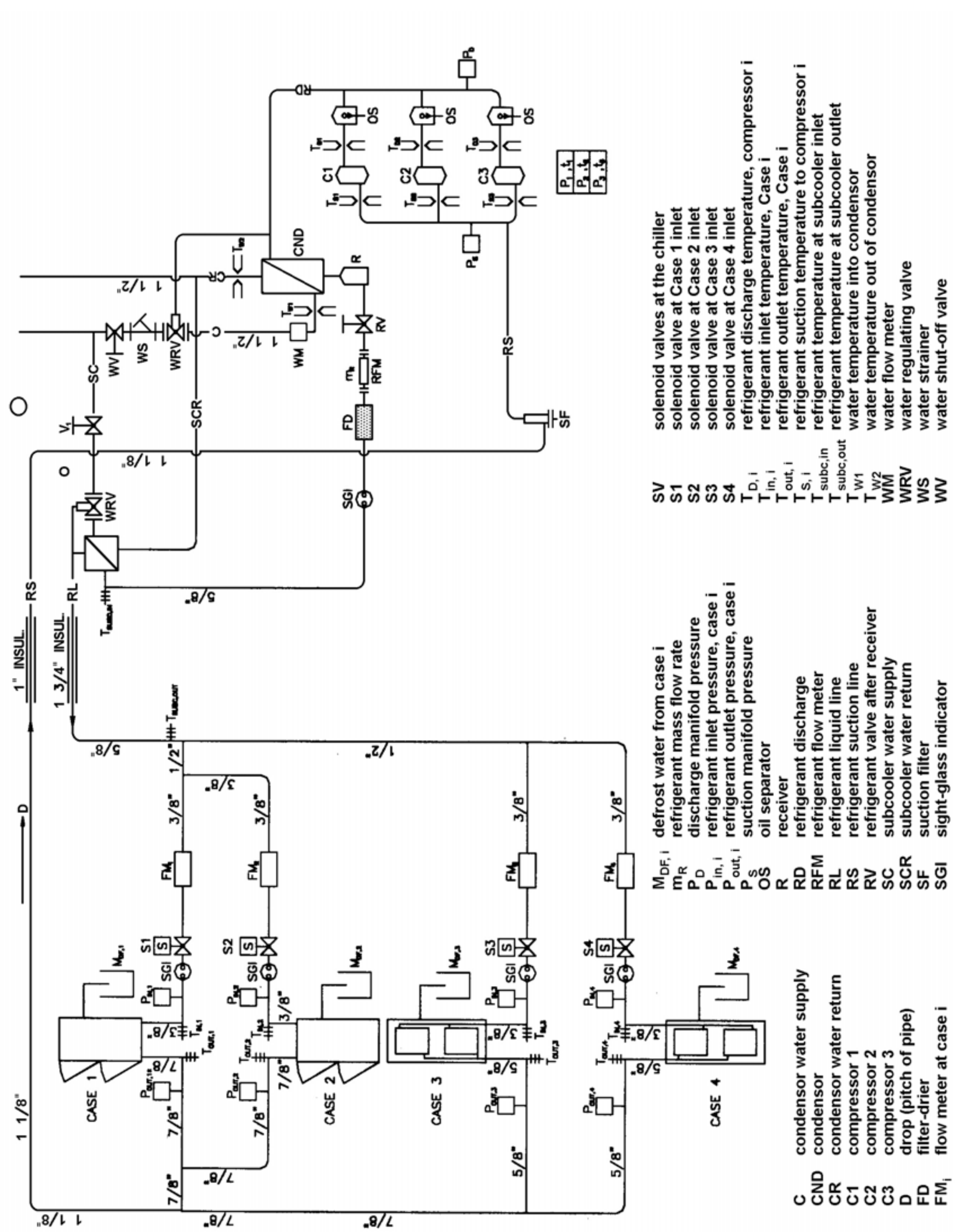

Figure 1. Schematic of Direct Expansion Test Rig

System Control

The refrigeration test rig is capable of operating the cases under either pressure or temperature control. Under pressure control, suction pressure at the manifold is specified, and the solenoid valves at the cases remain constantly open. The compressors cycle to maintain the desired suction pressure.

Under temperature control, solenoid valves at the cases open or close as needed to maintain discharge air temperature in the case. Suction pressure at the manifold is still maintained by compressor cycling. The original control strategy held the valves open for 90 seconds. Then the valves shut, but the discharge air was immediately sampled for set point. If the set point temperature was not achieved, the valves would reopen for another 90 seconds.

After several tests had been performed under temperature control, the MXV manufacturer believed that the original control strategy was detrimental to the operation of their valve. They hypothesized that the constant closing and opening of the solenoid valves disrupted the flow pattern which the MXV was establishing in the coil. At their suggestion, Ranco ETC microprocessor temperature controllers were installed on the cases. These controllers operated with a set point and a deadband which was set for +3 R at the recommendation of the MXV manufacturer.

For each case, defrost control involved setting defrost start time, defrost termination temperature, and default duration. Default duration sets the maximum length of defrost on-time should the termination temperature not be reached. Defrost start times for the cases were staggered by two hours after the first case initiated defrost. Reach-in cases were set for temperature termination at 48 °F with default duration of 45 minutes. For the open cases, termination occurred at 53 °F with a 60-minute default duration. Under the original configuration, each case defrosts once per 24 hours. At the request of the MXV manufacturer, some tests with the MXV assembly were performed with one defrost per 48 hours.

Data Collection

An automated data acquisition system collects and logs over 300 parameters once a minute. A running log of 36 hours of data is maintained and downloaded once every 24 hours. These instantaneous data are processed to calculate averages and 24-hour cumulative values. Figure 2 shows an example of the 36 hour data. A test period covers 27 hours—from the start of the first defrost of the first case to the start of the second defrost of the last case. The instantaneous average values of the test packages are combined over 24 hours to produce an integrated-average temperature (IAT) for each case. System and compressor energy data are calculated across the defrost and running cycle for the first case as shown in Figure 2. IATs and energy data for the individual cases are calculated across the defrost and running cycle for the individual case. Temperature and pressure data for the individual cases are averaged over the last three quarters of the running cycle for each individual case. Steady state operation is achieved when two days of data yields IATs with differences less than 0.5 R.

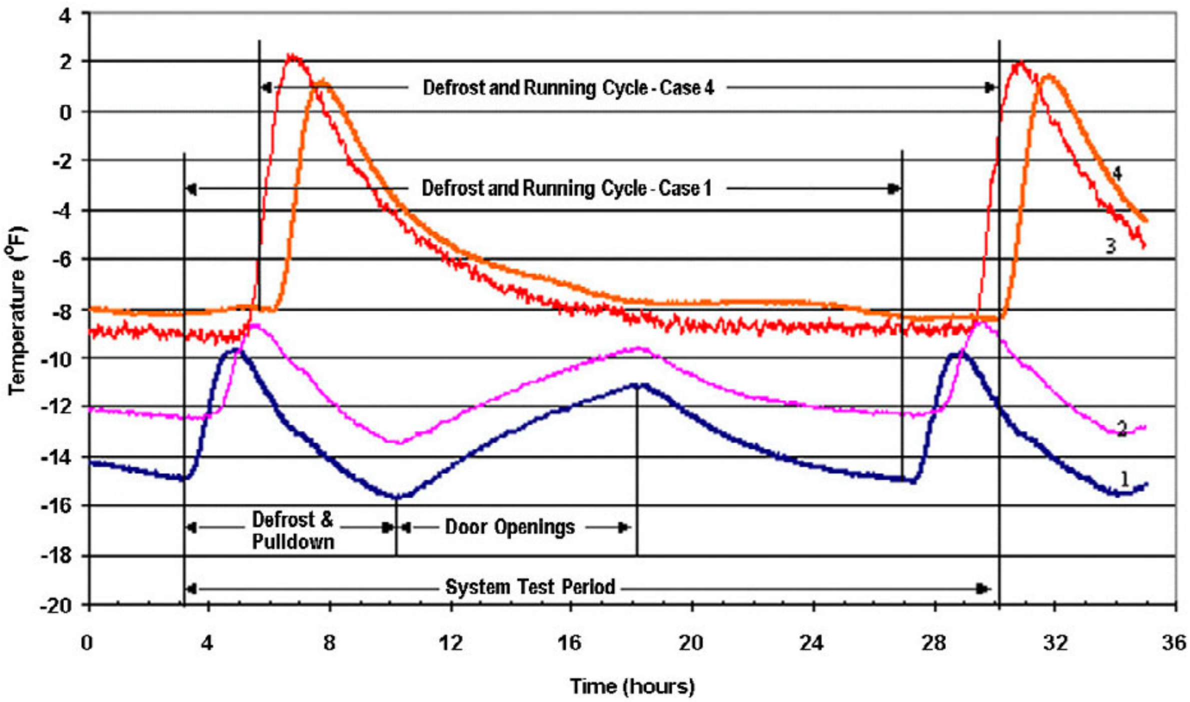

Figure 2. Average Package Temperatures over a 36 Hour Test Cycle for Reach-in Cases 1 & 2 and Open Cases 3 & 4

MXV Installation and Superheat Adjustment

MXV assemblies were installed on all four cases with the cooperation of the manufacturer. The original ¼ ton thermostatic expansion valves (TXV) were replaced with MXV assemblies. Each assembly included sixteen feet of ¼″ tubing at the end of the liquid line to each coil, the XTC valve, and new TXVs[2]. The capacity of the new TXVs could be altered by changing the expansion cartridge. For the open cases, the selected cartridges provide 0.14 tons of refrigeration at 67% of capacity, and the selected cartridges for the reach-in cases provide 0.44 tons of refrigeration at 80% of capacity. New nozzles were also placed on each distributor to the reach-in case coils. Figure 3 shows a photo of the installed XTC, the new thermostatic expansion valve, and the additional liquid line. Thermocouples were placed on the surface of two representative packages each in one reach-in and one open case.

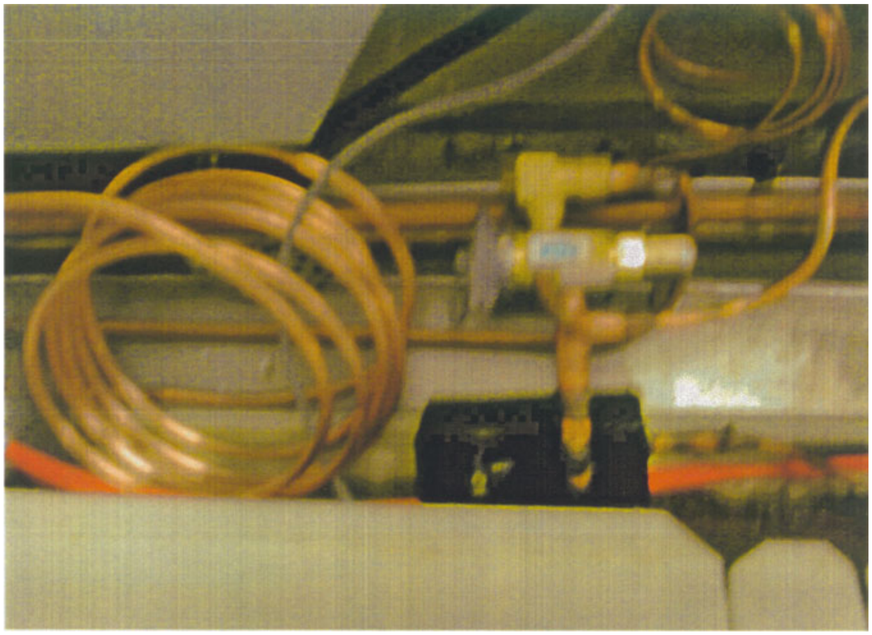

Figure 3. Multifunctional Valve Assembly

Recommended operating requirements for the MXV assembly are that coil superheats are between 0 and 5 R. After the installation, the manufacturer spent several days adjusting the superheats to their desired operation while the cases were running under temperature control. After the manufacturer left, it was noted that compressors were frosting heavily and were noisy, raising concern that liquid slugging was occurring. The system was shut down to protect the compressors. After restarting the system, several additional days were spent making adjustments to the reach-in cases in consultation with the MXV manufacturer. No progress was made to eliminate the frosting, and it was noted that the suction manifold temperature was approximately the same as the evaporating temperature.

Since the solenoid valve cycling on the cases made it difficult to adjust the superheats, it was decided to switch the system to pressure control. Once the system had pulled down, data were collected to observe the superheat settings. Figures 4 and 5 show the coil operation based on adjustments made under temperature control. The reach-in case data of Figure 4 show that there was approximately 24° of superheat between the inlet to the coil (Tx_in) and the outlet from the coil (Tx_out). An additional 10° of superheat was picked up in the suction

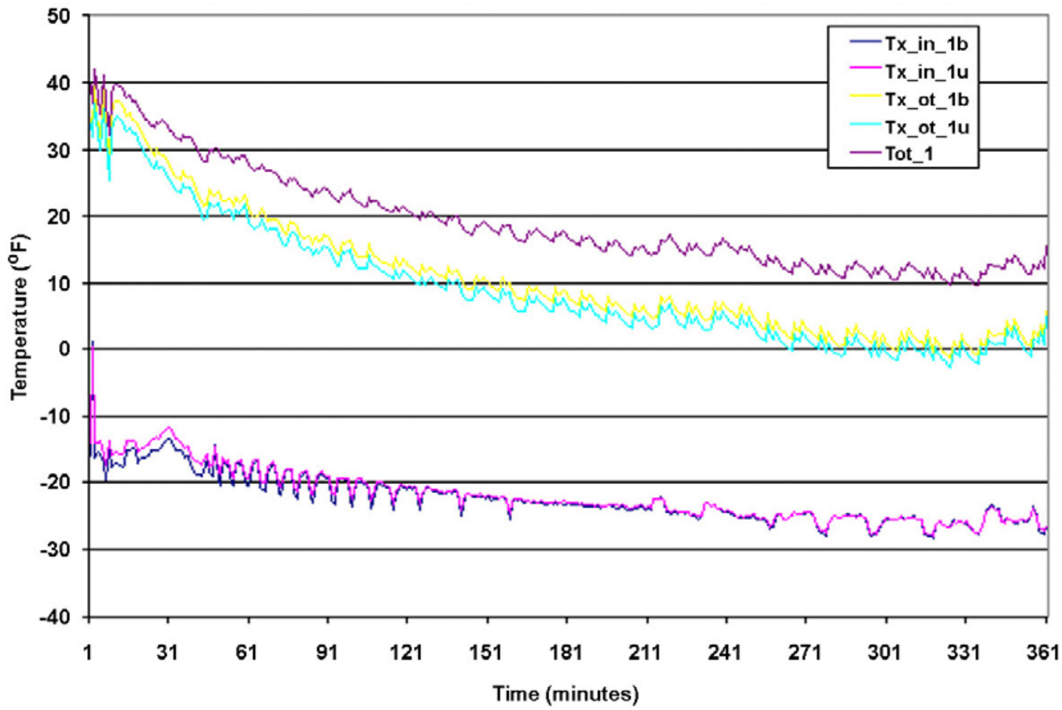

Figure 4. Reach-in Case 1 Coil Operation (6/28/01) Superheat Results when Adjustments Attempted During Solenoid Cycling

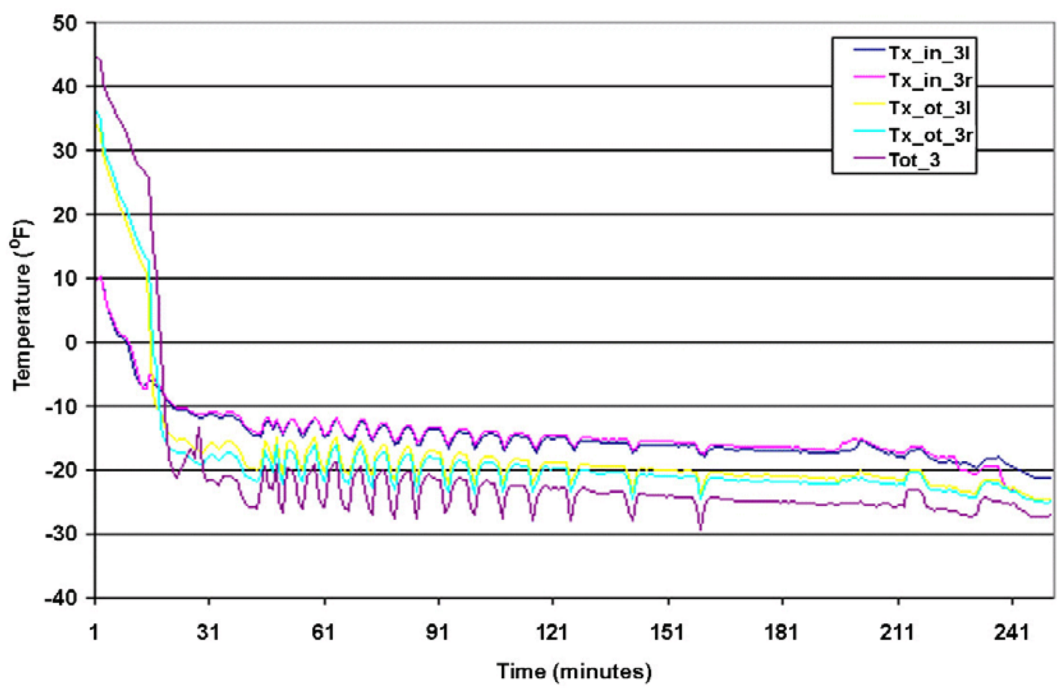

Figure 5. Open Case 3 Coil Operation (6/28/01) Superheat Results when Adjustments Attempted During Solenoid Cycling

line heat exchanger before the case outlet (Tout_1). For the open case data in Figure 5, temperatures for both coil outlet (Tx_out) and case outlet (Tout_3) were lower than the inlet temperature, indicating a significant liquid carryover from the open cases.

Figures 4 and 5 demonstrate the problems that can occur when trying to adjust expansion valves in systems with solenoid valves that cycle refrigerant flow on and off. Accurate readings of the superheat could only be measured when the system controller was set to operate under pressure control (no solenoid cycling). Then it became clear that the open cases were so flooded that liquid refrigerant was being carried over to the compressor. In addition, it was found that superheats in the reach-ins were 20 R higher than the desired setting. While in pressure control operation, all cases were adjusted to yield 5° of superheat at the coil outlet, which raised the suction manifold temperature from -25 °F to +25 °F and eliminated the compressor frosting. Data were sent to the MXV manufacturer showing the operation of coils before and after these adjustments. The final adjustment of the valves was accepted by the MXV manufacturer because all parties were in agreement that sufficient superheat is necessary to prevent flooding of the compressor and to ensure compressor safety during operation. Once testing started, no liquid slugging was observed, and compressor safety was maintained.

Test Matrix

The MXV assembly was evaluated at the conditions shown in Table 2.

Table 2. Test Matrix for MXV Assembly

System Control	24 Hour Defrost	48 Hour Defrost
Pressure	-30/75 °F[a]	-30/75 °F
Temperature	-30/75 °F	-30/75 °F
Ranco Temperature	-30/75 °F	no test
	-27/75 °F	no test
	-27/105 °F	-27/105 °F

[a] evaporating temperature/condensing temperature

After completion of the MXV tests, some additional tests under baseline configuration were performed in order to have equivalent baseline conditions for comparisons. These tests included the -27 °F evaporating conditions and tests with 48 hour defrost schedules.

Results and Discussion

In this section, results from the MXV tests are presented and compared to the baseline system. The prefix designations used in the summary tables are BS for baseline system tests and X for tests with the full MXV assembly. The summary tables in this section report averages of test replicates. Detailed tables of the results are available in Appendix A.

Pressure Control Tests

Several MXV tests were performed under pressure control with the superheat settings at conditions agreed to by the manufacturer. These data were averaged, and the results are shown in Table 3 (and Appendix Table Al). MXV test results were averaged and designated as XPC1. Baseline system tests are designated BSPC1 and BSPC2. The difference in energy consumption for the baseline system tests is due to differences in product temperatures and superheats. (Lower product temperatures reached in BSPC2 require more energy.)

Table 3. Comparison of Energy Use and Package Temperatures for Tests under Pressure Control at -30 °F Evaporating and 75 °F Condensing with One Defrost per 24 Hours.

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
BSPC1	-29.8	28.6	75.2	7.8	97.2	56.2	-14.1	-12.8	-6.7	-8.3
BSPC2	-29.7	12.0	74.7	8.3	104.2	63.6	-15.9	-14.1	-6.9	-8.8
XPC1	-30.2	29.5	75.1	7.7	101.2	60.1	-15.5	-14.9	-8.2	-9.9

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

Energy consumption in the baseline system tests with typical superheats (BSPC1) is about 3% lower than for the MXV system (XPC1). However, package temperatures for the MXV are about 1.5 R lower than the baseline. In baseline system tests with lower superheats (BSPC2), package temperatures approach those for the MXV, but energy usage is about 3% higher.

The reduced temperatures for the MXV system are a result of the reduced superheat settings, as evidenced in Figure 6. In earlier baseline studies under pressure control, package temp-

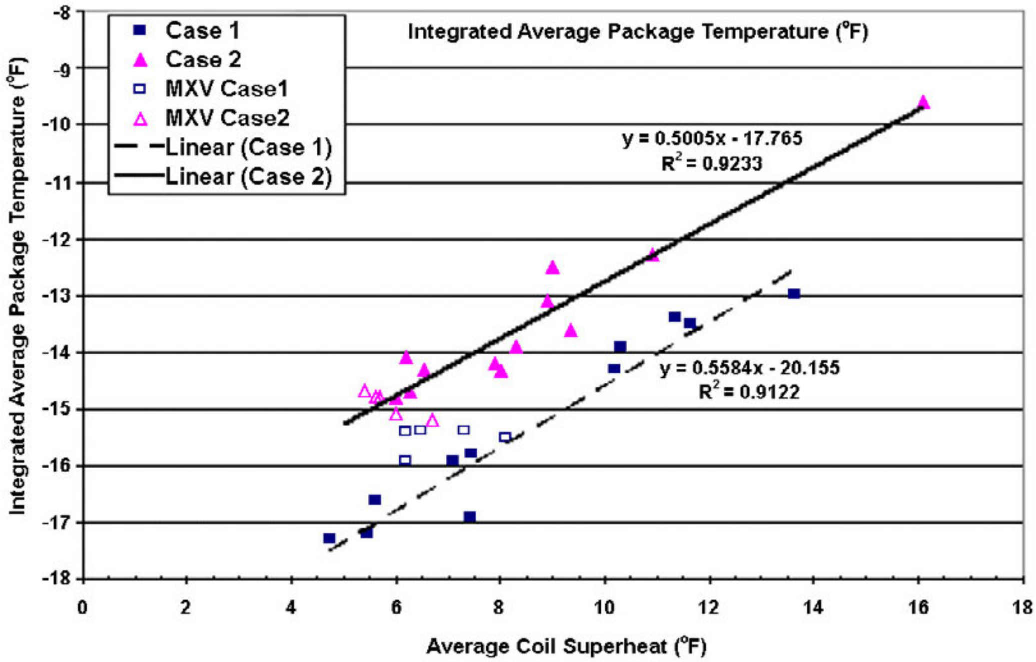

Figure 6. Effect of Superheat on Integrated Average Package Temperature for Reach-in Cases at T_{evap} = -30 °F under Pressure Control

eratures were found to be dependent on the superheat. In Figure 6, the tests of the MXV system under pressure control have been included with the earlier baseline work, and the results follow the same trend lines. MXV tests were also performed with one defrost per 48 hours. These results are shown in Table 4 (and Appendix Table A2). In the first 24 hours without a defrost (Day 1), energy use and package temperatures are both lower than in the 24 hour defrost schedule. In the 24 hour period which includes the defrost (Day 2), energy use and product temperatures are similar to the values observed under a 24 hour defrost schedule (XPC1 in Table 3).

Figures 7 and 8 show a comparison of the package temperatures over a 60 hour cycle for both 48 hour and 24 hour defrost schedule. The elimination of one defrost has no significant impact on these temperatures. XPC2Avg in Table 4 is the 24 hour average for the two day test. It shows that the packages are slightly colder under a 48 hour defrost with about a 4% reduction in energy use compared to a 24 hour defrost schedule. No baseline tests were performed with a 48 hour defrost schedule under pressure control.

Table 4. MXV Tests under Pressure Control at -30 °F Evaporating and 75 °F Condensing with One Defrost per 48 Hours.

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
XPC2Day1	-30.2	-30.6	75.1	8.0	94.25	57.03	-15.7	-15.4	-9.7	-12.7
XPC2Day2	-30.2	30.6	75.1	8.0	99.76	57.91	-15.0	-14.8	-7.9	-9.9
XPC2Avg	-30.2	30.6	75.1	8.0	97.01	57.47	-15.4	-15.1	-8.8	-11.3

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

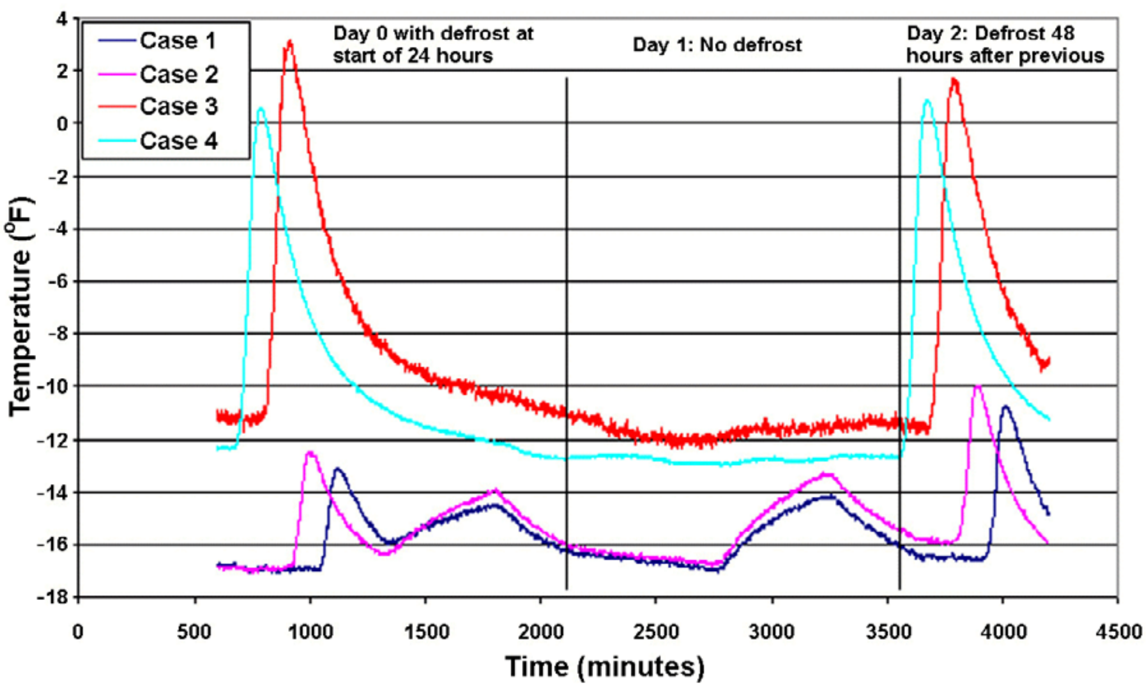

Figure 7. Packages over a 60 Hour Period (7/25/01–7/27/01) with One Defrost per 48 Hours. System under Pressure Control at -30 °F Evaporating and 75 °F Condensing

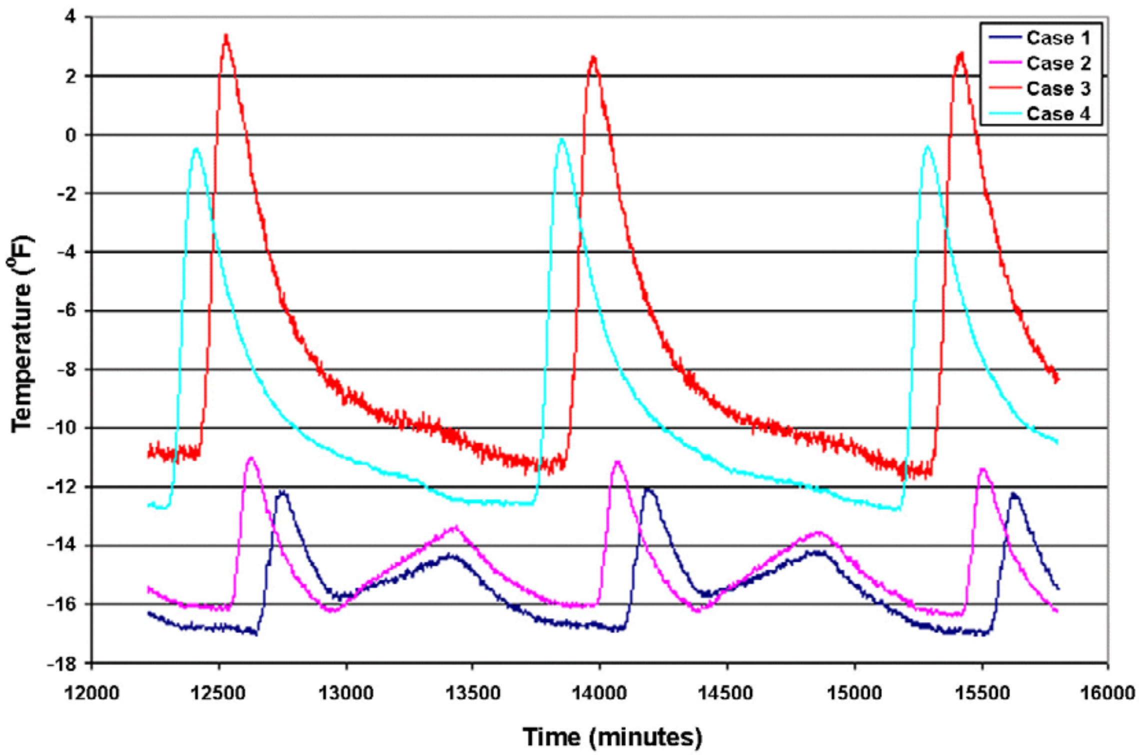

Figure 8. Packages over a 60 Hour Period (7/4/01–7/6/01) with One Defrost per 24 Hours. System under Pressure Control at -30 °F Evaporating and 75 °F Condensing.

Temperature Control Tests

In baseline tests under temperature control, set-points were chosen to give package temperatures of -10 °F in the reach-in cases and -6 °F in the open cases. These same package temperatures were matched with the MXV by appropriate setpoint selection. Table 5 (and Table Al in the Appendix) compares the temperature control tests of the baseline and MXV tests. Under the original temperature control strategy of the test rig, there is no significant difference in package temperatures and energy consumption between the baseline (BSTC1) and MXV (XTC1) tests.

As mentioned in the previous section, the MXV manufacturer was concerned that the original temperature control strategy did not allow the MXV assemblies to perform as designed. New temperature controllers were installed, and the MXV tests were repeated. As shown by XTC2, there was still no difference observed between the baseline test results and the MXV results.

Table 5. Comparison of Energy Use and Package Temperatures for Tests under Temperature Control at -30 °F Evaporating and 75 °F Condensing with one Defrost per 24 Hours

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
BSTC1	-29.8	32.7	74.6	9.7	97.9	57.5	-10.2	-10.1	-5.9	-7.1
XTC1	-30.2	34.1	74.9	12.7	97.0	56.5	-10.2	-9.7	-6.3	-6.7
XPC2	-29.5	33.4	75.4	8.4	98.9	57.7	-10.4	-10.2	-6.0	-6.4

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

During the temperature control tests, differences were noted in the pull-down times until the controllers start to cycle the case solenoids. Examples are shown in Figures 9 and 10. Figure 9 shows the refrigerant mass flow to Case 2 for a 70-minute period after defrost in the system with the MXV valve assembly. There is a 32-minute pull-down before the case solenoids begin cycling. Figure 10 shows the same parameter in the baseline system. Here the pull-down time is about 59 minutes. Although the MXV pull-down time is significantly shorter than the pull-down time for the baseline, this did not translate into lower energy consumption. This is because the 27 minutes of “saved” pull-down time became 27 minutes of compressor cycling time that was not a significant enough difference in energy consumption when incorporated over 24 hours of operation.

An example of the surface temperatures of the packages is shown in Figure 11 for two packages in Reach-in Case 1. Package 7 is near the front of the case at a relatively warm position, and package 36 is in the back on the bottom row at a relatively cold position. Two observations can be made from this figure. The first is that, for each package, there is no significant difference in the surface temperatures between the baseline system tests and the MXV tests. The second is that the pull-down times after defrost for the packages are comparable for both system tests. Therefore, the faster pull-down time for the coil to start the solenoid cycling that was observed in the MXV tests does not translate into a faster pull-down time for the packages.

Temperature control tests were also performed with a 48-hour defrost schedule. The MXV results are compared with the baseline system in Table 6. Comparing to results on a 24- hour

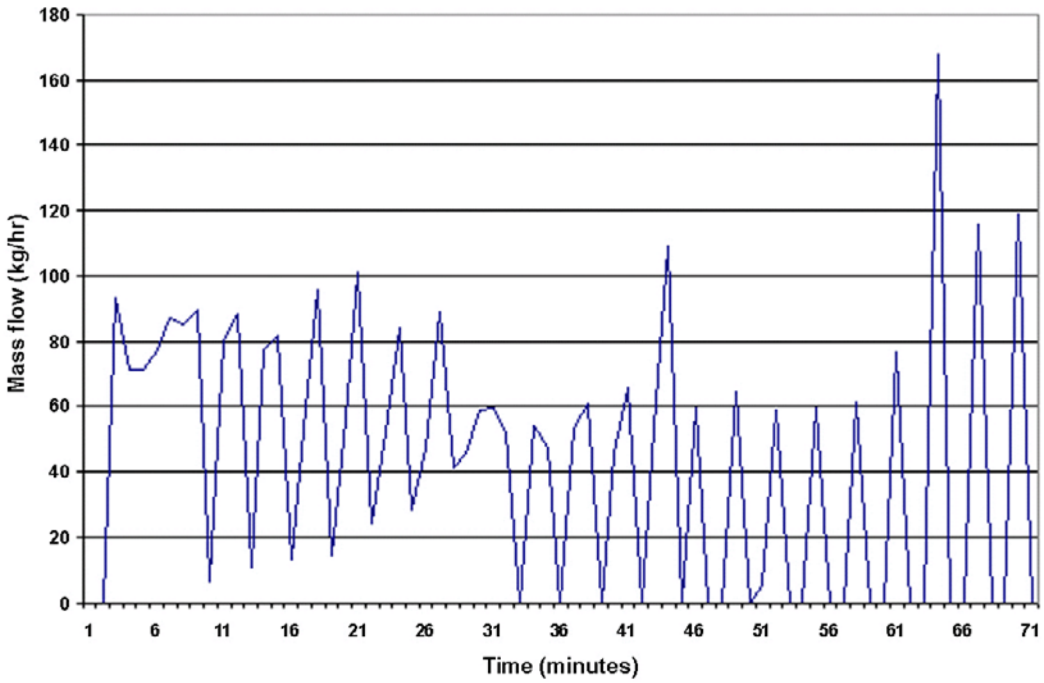

Figure 9. Refrigerant Flow (7/16/01) in Case 2 After Defrost with MXV Assembly. Evaporating Temperature = -30 °F, Condensing Temperature = 75 °F.

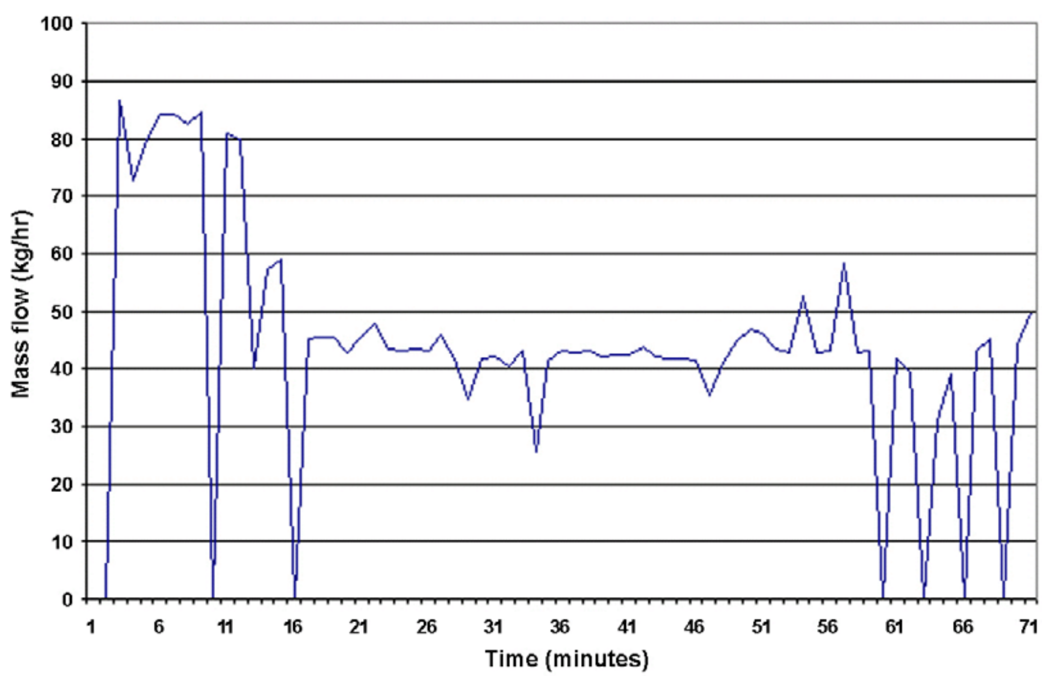

Figure 10. Refrigerant Flow (5/18/01) in Case 2 After Defrost with Baseline System. Evaporating Temperature = -30 °F, Condensing Temperature = 75 °F.

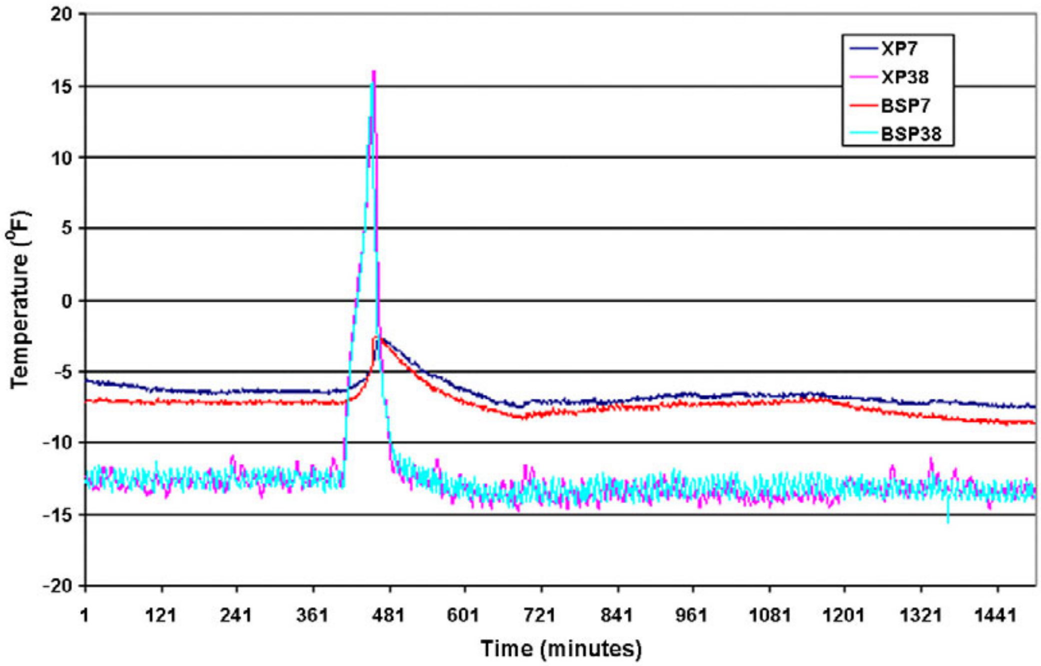

Figure 11. Surface Temperatures of Two Packages in Reach-in Case 1 during a 24 hour Period Including Defrost

Table 6. Comparison of Energy Use and Package Temperatures for Tests under Temperature Control at -30 °F Evaporating and 75 °F Condensing with one Defrost per 48 Hours

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
BSTC3Day1	-30.0	32.8	75.1	10.9	89.66	52.28	-10.5	-10.3	-8.5	-8.8
BSTC3Day2	-30.0	33.1	75.2	11.1	98.31	56.75	-10.2	-10.1	-6.2	-6.5
BSTC3Avg	-30.0	33.0	75.2	11.0	93.99	54.52	-10.4	-10.2	-7.4	-7.7
XPC2Day1	-30.3	36.8	75.1	12.4	89.10	52.50	-10.3	-9.6	-9.3	-9.1
XPC2Day2	-30..4	35.5	75.2	12.2	97.81	56.80	-10.1	-9.4	-6.1	-6.4
XPC2Avg	-30.4	36.2	75.2	12.3	93.46	54.65	-10.2	-9.5	-7.7	-7.8

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

defrost schedule (Table 5), energy savings and colder packages are achieved on the day without the defrost (Day 1) for both systems. On the day of the defrost (Day 2), energy consumption and package temperatures are similar to days with 24-hour defrost schedules for both systems. When the two days are averaged to give a 24-hour mean, both systems yield about a 4% energy savings per day when the number of defrosts is decreased to once per 48-hours with no significant impact on package temperature. Comparing the baseline and MXV systems for Day 1, Day 2, and the average, there is no significant difference in energy use.

Higher Suction Pressure

The MXV manufacturer suggested that, since the MXV tests under pressure control showed lower product temperature than the baseline system, the suction pressure could be raised in the temperature control tests with no degradation of product temperature but with some energy savings. Suction pressure was raised by 2 psi to give a nominal evaporating temperature of -27 °F. These results are shown in Table 7 (and Table A4 in Appendix A).

Baseline system tests were also performed at the higher suction pressure. As shown in Table 7, energy savings for both systems are about 4% over tests at -30 °F evaporating (Table 5), but again there is no difference in energy consumption between baseline and MXV. Product temperatures for the baseline system were less impacted by the increase in suction pressure than the MXV tests.

Table 7. Comparison of Energy Use and Package Temperatures for Tests under Temperature Control at -27 °F Evaporating and 75 °F Condensing with One Defrost per 24 Hours

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
BSTC4	-27.0	31.6	74.7	7.4	94.29	53.03	-10.4	-10.1	-5.1	-6.3
XPC4	-27.2	36.7	75.4	8.1	94.26	53.70	-9.8	-9.8	-5.1	-5.6

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

After completion of the tests at -27 °F evaporating, it was noted that, for the MXV tests of August 9 and part of August 10, the compressor controller appeared to be operating incorrectly by calling for the larger compressor to cycle on rather than to call for the smaller compressor first and then to switch to the larger compressor if the smaller could not meet the load. However for the remainder of August 10 and for all of August 11 the controller operated as expected by calling on the smaller compressor first. Despite this irregular operation, there appears to be no energy penalty to the MXV tests since energy use for each of the three days are comparable (see Table A11). The close result between the two types of operation is due to the controller cycling the compressors to maintain pressure set-point. To explain more fully, on August 11, when the controller was selecting the smaller compressor from start- up, the smaller compressor ran for an additional 721 minutes in 24 hours compared to its run time on August 9 and 10. During this same time period, the larger compressor ran 545 minutes (in 24 hours) less than its run time on August 9 and 10. In terms of daily energy consumption, the longer run time and lower wattage for the smaller compressor on August 11 was comparable to the shorter run time and higher wattage of the a larger compressor on August 9 and 10. Therefore, the combination of compressor wattage and run time yielded comparable total compressor daily energy use for all three tests.

Even though this analysis shows that the MXV system was not penalized, all other MXV tests conditions were reviewed to check whether the irregular operation had occurred at other conditions. None were found. Discussions with the manufacturer of the controller yielded no insights as to the possible cause of the irregular operation.

Higher Condensing Temperature

Tests were performed at 105 °F condensing at the request of the MXV manufacturer. Results comparing the baseline system and MXV assembly system are shown in Table 8. There is no difference in energy use between the systems; however, the baseline system had slightly lower product temperature for Case 4. For both systems, energy use is about 30% higher than their corresponding tests at 75 °F condensing.

Table 8. Comparison of Energy Consumption and Package Temperatures for Tests under Temperature Control at -27 °F Evaporating and 105 °F Condensing with one Defrost per 24 Hours

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
BSTC5	-26.8	29.3	105.2	11.0	125.80	84.34	-10.5	-10.4	-4.4	-6.7
XPC5	-26.7	-34.4	104.9	-10.2	125.49	84.41	-10.2	-10.0	-4.6	-5.8

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

A final MXV test was performed with a 48 hour defrost schedule at this higher condensing temperature. Results are shown in Table 9 (and Appendix A Table A2). There is an average 2.5% savings per day with the elimination of one defrost in 48 hours (compare to XTC5). No comparison is made to the baseline system since no baseline tests were performed at these conditions.

Table 9. MXV Tests under Temperature Control at -27 °F Evaporating and 105 °F Condensing with one Defrost per 48 Hours

Test	Te[a] °F	Ts[b] °F	Tc[c] °F	Sc[d] R	Energy		Package Temperatures			
					Total kWh/day	Comp. kWh/day	Case1 °F	Case 2 °F	Case 3 °F	Case 4 °F
XTC6Day1	-26.7	34.3	104.8	10.8	118.16	81.18	-10.4	-10.1	-7.6	-8.1
XTC6Day2	-26.7	35.0	104.8	10.1	126.26	84.53	-9.9	-9.8	-5.1	-5.7
XTC6Avg	-26.7	34.7	104.8	10.5	122.21	82.86	-10.2	-10.0	-6.4	-6.9

[a] evaporating temperature
[b] suction manifold temperature
[c] condensing temperature
[d] subcooling

Conclusions

Tests were performed at seven dfferent conditions on the MXV valve assembly. Baseline tests were performed at five of these conditions. Several of the test conditions were requested by the MXV manufacturer as conditions which should be able to show any benefit from the MXV assembly.

For tests performed with a 24 hour defrost schedule, the following conclusions are made:

- Under pressure control, the MXV assembly uses 3% more energy than the baseline; however, package temperatures are about 1.5 R lower.
- The lower package temperatures observed in the pressure control tests are a result of the lower superheats in the MXV assembly tests.
- Under temperature control with the original controllers, energy consumption of the baseline and MXV assembly systems are within 1% of one another, and package temperatures are similar.
- With the MXV assembly tested under an alternate temperature controller, energy consumption of the two systems are again within 1% of one another, and package temperatures are similar.
- Under temperature control, the coil pull-down time after defrost was about 50% shorter in the MXV assembly tests than in the baseline tests. When analyzed over the 24-hour test period, this did not yield a measurable energy difference because the saved pull-down time was converted to compressor cycling operation.
- Under temperature control, higher suction pressures can be used on both systems with

an associated savings of about 4% from the lower suction pressure tests.

- Package temperatures at higher suction pressures (e.g., -27 °F evaporating) for the baseline system were slightly lower than those during the MXV assembly tests although energy use was comparable.
- Energy use under temperature control for both systems is about 30% higher at 105 °F condensing compared to 75 °F condensing.
- At 105 °F condensing, energy consumption and package temperatures of the MXV assembly system are comparable to the baseline system.

For tests performed with a 48 hour defrost schedule the following conclusions are made:

- The average energy consumption in the tests with 48-defrost schedule is the same for both baseline and MXV systems and is 4% lower compared to tests with 24 hour defrosts schedule.
- The required defrost energy was higher than for a 24 hour defrost schedule due to the delay of the defrost; however, the resulting increase to the package temperatures was small.

References

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2. XTC product literature, XDX, Arlington Heights, IL 2000.

3. Standard 72-1998, “Method of Testing Open Refrigerators,” American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1998.

4. Standard 117-1992, “Method of Testing Closed Refrigerators,” American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1992.

Appendix A

Detailed Comparison Tables
and Individual Data Tables

Table A1. Comparison of Baseline System and MXV Assembly System with 24 hour Defrost Cycle under Temperature and Pressure Control at -30 °F Nominal Evaporating and 75 °F Nominal Condensing[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
Pressure Control																										
BSPC1[b]	75.7	64.5	74.8	65.6	-29.8	28.6	75.2	7.8	97.2	56.2	-14.1	10.3	31	905	-12.8	9.0	31	549	-6.7	10.0	53	1709	-8.3	10.3	43	1311
BSPC2[c]	75.6	63.6	74.9	67.0	-29.7	12.0	74.7	8.3	104.2	63.6	-15.9	6.2	31	1205	-14.1	5.4	36	1210	-6.9	10.0	54	1050	-8.8	10.7	43	1364
XPC1[d]	74.2	66.1	74.1	66.2	-30.2	29.5	75.1	7.7	101.2	60.1	-15.5	6.9	35	715	-14.9	5.9	36	980	-8.2	3.4	59	1458	-9.9	5.2	53	1251
Temperature Control																										
BSTC1[e]	74.4	63.9	74.6	65.3	-29.8	32.7	74.6	9.7	97.9	57.5	-10.2	16.0	33	1220	-10.1	12.5	31	1148	-5.9	11.9	60	1782	-7.1	14.9	52	1405
XTC1[f]	73.9	64.6	74.5	64.4	-30.2	34.1	74.9	12.7	97.0	56.5	-10.2	17.3	31	1147	-9.7	16.2	31	1087	-6.3	10.1	59	1591	-6.7	13.2	49	1381
XTC2[g]	74.9	66.1	74.2	64.4	-29.5	33.4	75.4	8.4	98.9	57.7	-10.4	13.6	33	856	-10.2	12.1	33	1159	-6.0	8.9	60	1753	-6.4	11.0	51	1627

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass
[b] Two tests performed in mid-March with high superheats on all cases
[c] Two tests performed in mid-March with low superheats on Cases 1 & 2 and high superheats on cases 3 &4
[d] Five tests performed in July with MXV assemblies and low superheats on all cases
[e] Three tests performed in mid-May
[f] Three tests performed in mid-July with MXV assemblies
[g] Two tests performed in August with MXV assemblies and alternate temperature control

Table A2. Summary of MXV Tests with 48 Hour Defrost Cycle.[a] Two 24-Hour Averages in Each Test Set, No Defrost on Day 1, One Defrost on Day 2, Days 1 and 2 Averaged to Get Average Daily Values.

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
Pressure Control (average of 2 tests in July and August at nominally -30 °F evaporating and 75 °F condensing)																										
XPC2Day1	74.0	64.6	74.5	64.7	-30.2	30.6	75.1	8.0	94.25	57.03	-15.7	8.3	0	0	-15.4	6.8	0	0	-9.7	4.0	0	0	-12.7	7.0	0	0
XPC2Day2	74.2	64.6	74.5	64.7	-30.2	30.6	75.1	8.0	99.76	57.91	-15.0	8.2	45	1626	-14.8	6.8	45	2417	-7.9	5.0	60	3684	-9.9	7.0	60	2523
XPC2Avg	74.1	64.6	74.5	64.7	-30.2	30.6	75.1	8.0	97.01	57.47	-15.4	8.3	23	813	-15.1	6.8	23	1209	-8.8	4.5	30	1842	-11.3	7.0	30	1262
Temperature Control (single test set in July at nominally -30 °F evaporating and 75 °F condensing)																										
XTC3Day1	74.2	64.8	74.6	64.5	-30.3	36.8	75.1	12.4	89.10	52.50	-10.3	18.3	0	0	-9.6	17.9	0	0	-9.3	11.0	0	0	-9.1	14.0	0	0
XTC3Day2	73.8	64.5	74.5	64.5	-30.4	35.5	75.2	12.2	97.81	56.80	-10.1	17.5	45	2936	-9.4	16.2	41	2452	-6.1	10.5	60	2923	-6.4	14.0	57	2751
XTC3Avg	74.0	64.7	74.6	64.5	-30.4	36.2	75.2	12.3	93.46	54.65	-10.2	17.9	23	1468	-9.5	17.1	21	1226	-7.7	10.8	30	1462	-7.8	14.0	29	1376
Temperature Control (single test set in August at nominally -27 °F evaporating and 105 °F condensing)																										
XTC6Day1	73.9	63.6	74.3	64.3	-26.7	34.3	104.8	10.8	118.16	81.18	-10.4	11.8	0	0	-10.1	10.5	0	0	-7.6	8.0	0	0	-8.1	8.6	0	0
XTC6Day2	73.9	63.6	74.3	64.3	-26.7	35.0	104.8	10.1	126.26	84.53	-9.9	11.4	44	1930	-9.8	10.3	43	2418	-5.1	8.2	60	3338	-5.7	9.0	59	3307
XTC6Avg	73.9	63.6	74.3	64.3	-26.7	34.7	104.8	10.5	122.21	82.86	-10.2	11.6	22	965	-10.0	10.4	22	1209	-6.4	8.1	30	1669	-6.9	8.8	30	1654

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A3. Comparison of Baseline System and System with MXV Assemblies with 48 Hour Defrost Cycle under Temperature Control at Nominally -30 °F Evaporating and 75 °F Condensing.[a] Two 24-Hour Averages in Each Test Set, No Defrost on Day 1, One Defrost on Day 2, Days 1 and 2 Averaged to Get Average Daily Values.

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
Single test set performed in July with nominally 12 °F liquid subcooling																										
XPC3Day1	74.2	64.8	74.6	64.5	-30.3	36.8	75.1	12.4	89.10	52.50	-10.3	18.3	0	0	-9.6	17.9	0	0	-9.3	11.0	0	0	-9.1	14.0	0	0
XPC3Day2	73.8	64.5	74.5	64.5	-30.4	35.5	75.2	12.2	97.81	56.80	-10.1	17.5	45	2936	-9.4	16.2	41	2452	-6.1	10.5	60	2923	-6.4	14.0	57	2751
XTC3Avg	74.0	64.7	74.6	64.5	-30.4	36.2	75.2	12.3	93.46	54.65	-10.2	17.9	23	1468	-9.5	17.1	21	1226	-7.7	10.8	30	1462	-7.8	14.0	29	1376
Single test set performed in early October with nominally 8 °F liquid subcooling																										
BSTC2Day1	74.6	64.6	74.2	65.4	-29.9	32.3	75.2	7.8	91.79	54.59	-10.4	13.6	0	0	-10.1	13.3	0	0	-8.6	11.5	0	0	-8.8	15.8	0	0
BSTC2Day2	74.9	63.8	74.3	64.9	-30.0	32.5	75.2	7.8	98.87	56.82	-10.2	14.7	42	1903	-10.0	13.2	40	2467	-6.3	11.6	60	3391	-6.4	15.6	60	2654
BSTC2Avg	74.8	64.2	74.3	65.2	-30.0	32.4	75.2	7.8	95.33	55.71	-10.3	14.2	21	952	-10.1	13.3	20	1234	-7.5	11.6	30	1696	-7.6	15.7	30	1327
Single test set performed in mid-October with nominally 12 °F liquid subcooling																										
BSTC3Day1	74.1	65.4	74.7	65.3	-30.0	32.8	75.1	10.9	89.66	52.28	-10.5	14.4	0	0	-10.3	12.7	0	0	-8.5	11.7	0	0	-8.8	15.6	0	0
BSTC3Day2	74.1	65.6	74.7	65.7	-30.0	33.1	75.2	11.1	98.31	56.75	-10.2	14.7	40	2283	-10.1	12.9	41	2450	-6.2	11.3	59	3226	-6.5	15.5	59	2626
BSTC3Avg	74.1	65.5	74.7	65.5	-30.0	33.0	75.2	11.0	93.99	54.52	-10.4	14.6	20	1142	-10.2	12.8	21	1225	-7.4	11.5	30	1613	-7.7	15.6	30	1313

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A4. Comparison of Baseline System and MXV Assembly System with 24 hour Defrost Cycle under Temperature Control at -27 °F Nominal Evaporating and 75 °F Nominal Condensing[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
BSTC4[b]	74.2	64.5	74.7	65.6	-27.0	31.6	74.7	7.4	94.29	53.03	-10.4	11.1	30	721	-10.1	9.9	28	790	-5.1	10.5	55	1732	-6.3	13.0	52	1267
XTC4[c]	75.6	64.5	73.7	65.2	-27.2	36.7	75.4	8.1	94.85	53.70	-9.8	12.3	35	779	-9.8	11.1	35	926	-5.1	7.6	60	1661	-5.6	9.6	51	1676

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass
[b] Two tests performed in early October
[c] Three tests performed in mid-August

Table A5. Comparison of Baseline System and MXV Assembly System with 24 hour Defrost Cycle under Temperature Control at -27 °F Nominal Evaporating and 105 °F Nominal Condensing[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
BSTC5[b]	74.2	65.8	74.7	66.1	-26.8	29.3	105.2	11.0	125.8	84.3	-10.5	9.4	33	1120	-10.4	7.8	31	1038	-4.4	10.5	55	1821	-6.7	10.2	53	1271
XTC5[c]	73.9	64.1	74.3	64.8	-26.7	34.4	104.9	10.2	125.5	84.4	-10.2	11.3	34	1228	-10.0	10.3	35	1183	-4.6	8.1	60	1739	-5.8	7.5	53	1736

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass
[b] One test performed in mid-October
[c] One test performed in mid-August

Table A6. MXV Tests at -30 °F Nominal Evaporating and 75 °F Nominal Condensing under Pressure Control with 24 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
07/03/01	74.4	66.5	73.9	66.3	-30.1	28.0	75.1	7.2	101.89	60.49	-15.9	6.2	36	904	-14.7	5.4	37	1098	-8.4	2.8	58	–□	-10.2	3.9	51	1004
07/05/01	74.4	67.5	73.4	68.4	-30.1	29.2	75.1	7.8	101.27	60.19	-15.4	6.2	37	910	-14.8	5.6	38	1326	-8.0	2.9	60	1427	-9.9	4.5	53	1254
07/06/01	74.2	67.3	73.7	67.4	-30.2	29.3	75.1	7.6	99.14	58.19	-15.4	6.5	38	863	-14.8	5.7	37	1340	-8.2	3.0	58	1425	-9.8	4.7	53	1234
07/22/01	74.1	64.4	74.7	64.5	-30.2	29.5	75.0	8.1	100.51	59.44	-15.4	7.3	36	712	-15.1	6.0	36	1043	-8.2	4.0	60	1505	-9.8	6.4	54	1460
07/25/01	74.1	64.6	74.7	64.6	-30.2	31.6	75.3	8.0	103.00	61.95	-15.5	8.1	30	186	-15.2	6.7	32	95	-8.1	4.3	60	1475	-9.7	6.7	55	1304
XPC1	74.2	66.1	74.1	66.2	-30.2	29.5	75.1	7.7	101.16	60.05	-15.5	6.9	35	715	-14.9	5.9	36	980	-8.2	3.4	59	1458	-9.9	5.2	53	1251

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A7. MXV Tests at -30 °F Nominal Evaporating and 75 °F Nominal Condensing under Pressure Control with 48 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
07/26/01	73.9	64.5	74.7	64.6	-30.2	31.5	75.2	7.9	94.51	57.32	-15.8	8.2	0	0	-15.4	6.7	0	0	-11.7	4.2	0	0	-12.8	6.5	0	0
07/27/01	74.1	64.5	74.6	64.5	-30.2	30.5	75.1	7.8	98.40	56.65	-15.1	8.1	45	1583	-14.8	6.6	45	2590	-8.7	4.5	60	2795	-9.9	6.5	60	2582
Average	74.0	64.5	74.7	64.6	-30.2	31.0	75.2	7.9	96.46	56.99	-15.5	8.2	23	792	-15.1	6.7	23	1295	-10.2	4.4	30	1398	-11.4	6.5	30	1291
07/28/01	74.0	64.3	74.5	64.4	-30.3	30.8	75.0	7.1	95.01	54.74	-15.7	8.1	0	0	-15.5	6.5	0	0	-11.7	4.5	0	0	-12.7	6.1	0	0
07/29/01[b]	73.9	63.7	74.6	64.4	-25.2	31.0	74.6	6.6	92.58	50.94	-11.5	9.0	45	1460	-11.6	8.4	45	2273	-6.4	8.4	60	2796	-8.0	9.4	60	2530
Average[b]	74.0	64.0	74.6	64.4	-27.8	30.9	74.8	6.9	93.80	52.84	-13.6	8.6	23	730	-13.6	7.5	23	1137	-9.1	6.5	30	1398	-10.4	7.8	30	1265
08/01/01	74.1	64.4	74.4	64.9	-30.2	30.9	75.0	8.4	93.99	56.73	-15.6	8.3	0	0	-15.4	6.9	0	0	-7.7	3.8	0	0	-12.6	6.6	0	0
08/02/01	74.2	64.6	74.3	64.8	-30.1	30.6	75.0	8.2	101.12	59.17	-14.9	8.2	45	1668	-14.7	7.0	45	2244	-7.0	5.0	60	4572	-9.8	6.9	60	2464
Average	74.2	64.5	74.4	64.9	-30.2	30.8	75.0	8.3	97.56	57.95	-15.3	8.3	23	834	-15.1	7.0	23	1122	-7.4	4.4	30	2286	-11.2	6.8	30	1232
08/03/01	74.3	64.8	74.2	64.3	-30.1	31.0	75.0	8.2	95.20	57.87	-14.6	8.9	0	0	-15.1	7.3	0	0	-10.8	4.8	0	0	-12.4	6.8	0	0
AvgDay1:4[c]	74.1	64.5	74.5	64.6	-30.2	31.1	75.1	7.9	94.68	56.67	-15.4	8.4	0	0	-15.4	6.9	0	0	-10.5	4	0	0	-12.6	7	0	0
XPC2Day1[d]	74.0	64.5	74.6	64.8	-30.2	31.2	75.1	8.2	94.25	57.03	-15.7	8.3	0	0	-15.4	6.8	0	0	-9.7	4	0	0	-12.7	7	0	0
XPC2Day2[e]	74.2	64.1	74.5	64.7	-30.2	30.6	75.1	8.0	99.76	57.91	-15.0	8.2	45	1626	-14.8	6.8	45	2417	-7.9	5	60	3684	-9.9	7	60	2523

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

[b] Tests with two compressors off line; not used in averages

[c] Average of four Day 1 tests (07/26/01, 07/28/01, 08/01/01, 08/03/01)

[d] Average of two Day 1 tests (07/26/01, 08/01/01)

[e] Average of two Day 2 tests (07/27/01, 08/02/01)

Table A8. MXV Tests at -30 °F Nominal Evaporating and 75 °F Nominal Condensing under Temperature Control with 24 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
07/14/01	74.0	64.4	74.5	64.3	-30.1	34.4	74.9	12.4	97.07	56.22	-10.0	17.1	32	1122	-9.8	16.1	31	1111	-6.5	9.7	60	1648	-6.8	12.9	49	1375
07/15/01	74.0	64.4	74.5	64.4	-30.2	33.7	74.9	12.8	96.79	56.53	-10.0	17.3	31	1160	-9.7	16.1	30	1075	-6.2	10.2	58	1648	-6.7	13.0	49	1373
07/16/01	73.8	64.4	74.5	64.4	-30.2	34.1	75.0	12.9	97.24	56.67	-10.6	17.6	31	1159	-9.7	16.3	32	1075	-6.3	10.5	58	1476	-6.6	13.7	50	1395
XTC1	73.9	64.4	74.5	64.4	-30.2	34.1	74.9	12.7	97.03	56.47	-10.2	17.3	31	1147	-9.7	16.2	31	1087	-6.3	10.1	59	1591	-6.7	13.2	49	1381

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A9. MXV Tests at -30 °F Nominal Evaporating and 75 °F Nominal Condensing under Alternate Temperature Control with 24 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
08/06/01	74.6	66.6	74.3	64.3	-29.3	33.5	75.3	8.4	99.17	57.88	-10.3	13.6	32	919	-10.1	12.1	33	1185	-6.2	8.9	60	1825	-6.2	11.2	52	1702
08/07/01	75.1	65.5	74.1	64.5	-29.6	33.3	75.4	8.3	98.58	57.43	-10.4	13.6	34	793	-10.3	12.1	33	1133	-5.7	8.8	59	1680	-6.5	10.8	50	1552
XTC2	74.9	66.1	74.2	64.4	-29.5	33.4	75.4	8.4	98.88	57.66	-10.4	13.6	33	856	-10.2	12.1	33	1159	-6.0	8.9	60	1753	-6.4	11.0	51	1627

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A10. MXV Tests at -30 °F Nominal Evaporating and 75 °F Nominal Condensing under Temperature Control with 48 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
07/19/01	73.8	64.5	74.5	64.5	-30.4	35.5	75.2	12.2	97.81	56.80	-10.1	17.5	45	2936	-9.4	16.2	41	2452	-6.1	10.5	60	2923	-6.4	14.0	57	2751
07/20/01	74.2	64.8	74.6	64.5	-30.3	36.8	75.1	12.4	89.10	52.50	-10.3	18.3	0	0	-9.6	17.9	0	0	-9.3	11.0	0	0	-9.1	14.0	0	0
XTC3[b]	74.0	64.7	74.6	64.5	-30.4	36.2	75.2	12.3	93.46	54.65	-10.2	17.9	23	1468	-9.5	17.1	21	1226	-7.7	10.8	30	1462	-7.8	14.0	29	1376

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

[b] 24-hr average of two-day data

Table A11. MXV Tests at -27 °F Nominal Evaporating and 75 °F Nominal Condensing under Temperature Control with 24 hour Defrost Cycle[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
08/09/01	76.7	65.3	73.3	65.8	-27.1	39.2	75.1	8.5	93.70	52.61	-9.7	12.2	34	916	-9.7	11.0	36	1172	-5.4	7.5	60	1597	-5.9	9.2	51	1617
08/10/01	75.9	64.8	73.4	65.5	-27.1	36.5	75.5	7.8	93.76	52.74	-9.7	12.2	34	357	-9.8	10.8	33	383	-4.5	7.9	60	1667	-5.2	10.2	51	1701
08/11/01	74.1	63.8	74.3	64.2	-27.4	34.4	75.6	8.0	95.32	55.76	-10.1	12.6	36	1065	-10.0	11.4	36	1223	-5.3	7.4	60	1720	-5.8	9.5	52	1709
XTC4	75.6	64.6	73.7	65.2	-27.2	36.7	75.4	8.1	94.26	53.70	-9.8	12.3	35	779	-9.8	11.1	35	926	-5.1	7.6	60	1661	-5.6	9.6	51	1676

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

Table A12. MXV Tests at -27 °F Nominal Evaporating and 105 °F Nominal Condensing under Alternate Temperature Control[a]

Test	Zone 1		Zone 2		Te	Ts	Tc	SC	Energy		Case 1				Case 2				Case 3				Case 4			
	Dry °F	Wet °F	Dry °F	Wet °F					Total kWh/day	Comp. kWh/day	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g	IAT °F	SH R	Dt min	Mw g
08/17/01[b]	73.9	63.6	74.3	64.3	-26.7	34.3	104.8	10.8	118.16	81.18	-10.4	11.8	0	0	-10.1	10.5	0	0	-7.6	8.0	0	0	-8.1	8.6	0	0
08/18/01[c]	73.9	63.6	74.3	64.3	-26.7	35.0	104.8	10.1	126.26	84.53	-9.9	11.4	44	1930	-9.8	10.3	43	2418	-5.1	8.2	60	3338	-5.7	9.0	59	3307
XTC6[d]	73.9	63.6	74.3	64.3	-26.7	34.7	104.8	10.5	122.21	82.86	-10.2	11.6	22	965	-10	10.4	22	1209	-6.4	8.1	30	1669	-6.9	8.8	30	1654

[a] Nomenclature: BS= average of baseline system tests, X= average of system tests with MXV assembly, PC= pressure control, TC= temperature control, Dry= dry bulb temperature, Wet= wet bulb temperature, Te= evaporating temperature, Ts= suction manifold temperature, Tc= condensing temperature, SC= subcooling, Total= total energy used by system per day, Comp.= energy used by compressors per day, IAT integrated average package temperature, SH= superheat, Dt= duration of defrost, Mw= condensate water mass

[b] 24-hr defrost cycle

[c] 48-hr defrost cycle

[d] 24-hr average of two-day data

Appendix B

Letter from Manufacturer and Discussion

August 30, 2001

Mr. Georgi Kazachki
U.S. Environmental Protection Agency
Office of Research and Development
Air Pollution Prevention and Control Div., MD-63
Research Triangle Park, NC 27711

Dear Mr. Kazachki:

Thank you for taking the time out of your busy schedule to meet with us. I appreciate your desire and ability to collect data and to be diligent in your efforts.

As you heard last week, when the idea of testing at EPA was first brought up by DOE, I declined due to the perception that the testing would allow little input to adapt to the mode of testing, and that proprietary information might be shared with the industry. Our mode of testing had to that point been in compressor cycling and evaporative off time. At the time, I felt that without lengthy discussion, XDX® could not be tested at EPA. When interest was renewed at DOE for testing at EPA, as you recall, we tried to set-up a meeting to discuss the system prior to testing, but your pace was hurried due to scheduling and you wanted the system installed prior to any discussion with me. The spirit of adaptation that you conveyed is what compelled me to believe that testing at EPA would work out well and time proved this to be as such. You were all quite cooperative and allowed adaptation to the testing throughout the test as promised. We feel that "evaluation" of the test results clearly demonstrates a difference and that verification clearly shows that XDX® has benefits over a conventional set-up and operated system. You stated that the data did not demonstrate a difference. I hope you will have time to review the data in light of our discussion. I believe that upon this review you will concur with our observations.

You and Cynthia each stated during the meeting that the purpose of the test at EPA as requested by DOE was to evaluate and verify our XDX® system without expectation against the conventionally found system. Yet, factory specifications for the evaporative load, is how you have weighed work performed.

Adaptation of XDX® system to the branded parallel compressor rack system with new technology created significant problem that were not anticipated by any of us. I identify that the adaptation amidst the normal disruptions with a laboratory such as scheduling, power outages and deadlines provides some individual tests that cannot be averaged into the sum of tests due to many operational differences. Common sense would support that XDX® performance should be weighted by category of operational mode and not be from combination and averaging results as is commonly done since the modes of operation were so different. It is not common to combine results at different condensing temperatures, superheats or control modes, or varied types of controllers, as you know.

Our review of the data clearly shows:

- Product temperature reduction of 1 ½°F when operated with XDX® in pressure control mode was repeatedly shown.
- XDX® operated with the increased suction pressure test at a 4% kWh/Day savings.
- That the defrost reduction can save 4.5 kWh of electric heat each day that is operated without defrost.
- The data demonstrates at least 8 ½% reduction in kWh with XDX®. Before compressor normalizations (that should be applied after the realization that a reduction in load with XDX® higher suction pressure brought on a larger compressor because of the rack controller). XDX® can demonstrate additional kilowatt hour savings of at 75°F condensing temperature and a greater kilowatt hour savings under the more grueling 105°F condensing temperature in a simulation of summer peak demand.
- That pull down from post defrost in 22 minutes with XDX® as opposed to 58 Minutes with conventionally set-up and operated system as per the manufacturer, which reduced pull down time by 62% and pull down kWh from 4.17 to 1.86 for a savings of 55.4%.

- XDX® frost formation as videotaped shows less frost blockage than as conventional operated.
- A reduction in defrost requirement with XDX® to once every 48 hours (or longer) at once in 24 hours from conventional operated system.
- XDX® operated with an increased suction pressure with similar product temperature (.3°F is within the margin of error the sensor).
- That evaporative coil temperature can exceed supply air temperature by 1 to 5 degrees, supporting the theory of sublimation of a portion of the frost.
- That the XDX® operated evaporative coil can hold much more water than the conventionally sold and operated system and not impede performance.
- Reduced condenser load with XDX®.

We appreciate you experience and the diligence of your staff and facility. We appreciate Cynthia for her compilation of data and for her efforts toward impartiality. Her discussions in observation of supply air and evaporative coil temperatures and of where IATs (integrated average temperature) different prove to be helpful in the meeting. She should be commended for bringing us all to a smooth outcome.

In summary, the EPA conclusion to identify the test as raising unforeseen questions that leads to additional testing is a good idea

Your focus upon IATs is important as this the purpose for refrigeration. The energy data collected, which is often not done at the OEM or supermarket level should also be commended.

The data shows significant differences between XDX® and the conventionally operated system. The conclusion that we cannot operate in a supermarket is incompletely drawn given the issues related to controller logic and pulse thermostats.

We hope that the photographs of the different flow regimes in the clear evaporative coil were helpful. Let me know if you desire the video of this when it is available.

Your quest for understanding is admirable, but when assessing disruptive technologies like XDX® conventional experience and knowledge can often get in the way as one tries to explain away benefits, rather than analyzing data and seeing results. Please review the data in this new light. Thank you again and we will send additional observations as you requested.

Sincerely,

David Wightman
XDX® Innovative Refrigeration

Discussion of Manufacturer-stated Benefits

(Note: The list below is reproduced exactly as printed in the letter. Response is shown in italics.)

1. Product temperature reduction of 1 ½°F when operated with XDX® in pressure control mode was repeatedly shown. *Reduced package temperatures were noted in pressure control tests. (See Table 1.) However, this benefit came at a 3% energy penalty.*

2. XDX® operated with the increased suction pressure test at a 4% kWh/Day savings. *Similar savings were noted in the baseline system when suction pressure was increased to the same condition as the XDX test. (See Table 6.)*

3. That the defrost reduction can save 4.5 kWh of electric heat each day that is operated without defrost. *Identical savings are reached with the baseline system when it is operated without defrost. (See Table 5.)*

4. The data demonstrates at least 8 ½% reduction in kWh with XDX®. Before compressor normalizations (that should be applied after the realization that a reduction in load with XDX® higher suction pressure brought on a larger compressor because of the rack controller). XDX® can demonstrate additional kilowatt hour savings of at 75°F condensing temperature and a greater kilowatt hour savings under the more grueling 105°F condensing temperature in a simulation of summer peak demand. *Since there are no direct test comparisons which show this 8 1/2% reduction, it is assumed that this number comes from combining items 2 (increased suction pressure) and 3 (elimination of one defrost). The same additive reduction can be achieved with the baseline system since the same savings are noted on items 2 and 3. Energy savings at the 105 °F condensing temperature for the XDX tests are comparable to the baseline tests.*

5. That pull down from post defrost in 22 minutes with XDX® as opposed to 58 Minutes with conventionally set-up and operated system as per the manufacturer, which reduced pull down time by 62% and pull down kWh from 4.17 to 1.86 for a savings of 55.4%. *Since this statement does not specify which test dates or conditions these numbers are taken from, the discussion will address the numbers presented in Figures 9 and 10. (It should be noted that when this "savings" was presented by the XDX representatives at the close-out meeting, they stated that the energy numbers were taken from the differences in the <u>total system</u> energy usage during the associated pull-down periods. It was brought to their attention that these energy numbers would then reflect not only the energy used for pull-down of the example case but also the energy used by the other cases to maintain steady-state while the example case pulled-down.) In Figures 9 and 10, there is a 32-minute pull-down for the XDX and a 58-minute pull-down for the baseline system. The total system energy used during the XDX pull-down was 2.40 kWh and during the baseline pull-down was 4.28 kWh. However, this compares 32 minutes of usage to 58 minutes of usage. A correct comparison would be for the same time periods*

across both systems. The total system energy usage over 58 minutes in the XDX test is 4.21kWh compared to 4.28 kWh for the baseline system over the same period. This is a reduction of 0.07 kWh which is less than 0.1% of the energy used in 24 hours. And although the coil pulled down faster for the solenoid to start cycling, this did not translate to a faster pull-down of the packages as shown in Figure 12.

6. XDX® frost formation as videotaped shows less frost blockage than as conventional operated. *This is a subjective conclusion.*

7. A reduction in defrost requirement with XDX® to once every 48 hours (or longer) at once in 24 hours from conventional operated system. *The baseline (conventional) system also performed well with the defrost requirement reduced to once in 48 hours. (See Table 5.) Tests were not performed on either system to determine how much the defrost requirement could be reduced before package temperatures and performance degraded to unacceptable levels.*

8. XDX® operated with an increased suction pressure with similar product temperature (.3°F is within the margin of error the sensor). *The baseline system also operated at increased suction pressure with better agreement of the package temperatures than was observed with the XDX tests. (See Table 6.)*

9. That evaporative coil temperature can exceed supply air temperature by 1 to 5 degrees, supporting the theory of sublimation of a portion of the frost. *A review of the daily averages for air and coil temperatures does not yield any tests where coil temperature exceeds the supply air temperature. This is as expected since the air is the heat transfer medium between the cold coil and the product. Although instantaneous occurrences are not excluded, if they occurred to any significance, it would be detrimental to the product quality because of ice desublimation onto the product surface. Aside from this, a significant occurrence would have impacted energy usage through reduced ice formation on the coil, and thus reduced defrost demand and better heat transfer over the coil. No such energy reduction was observed compared to the baseline system.*

10. That the XDX® operated evaporative coil can hold much more water than the conventionally sold and operated system and not impede performance. *An objective evaluation of this can be made by comparing the defrost water collected with the XDX and baseline systems operating under the same conditions. As shown in Table A1 for the 24-hour defrost cycle tests BSTC1 and XTC1, the total defrost water collected is 5.56 kg for the baseline system and 5.21 kg for the XDX systems. As shown in Table A3 for the 48-hour defrost cycle tests BSTC3 and XTC3, the total defrost water collected is 10.59 kg for the baseline system and 11.06 kg for the XDX system. The mass differences in both these examples are well within the repeatability and accuracy limits of this parameter.*

11. Reduced condenser load with XDX®. *Condenser load is calculated by an energy balance on the water-side of the condenser. The following table shows the difference in the*

averaged loads of the XDX and baseline systems for all temperature control tests. Only in the 48 hour defrost test is the XDX condenser load lower than the baseline system.

Table B1. Comparison of Condenser Loads for Baseline and MXV Tests Performed under Temperature Control

Condition (Evap/Cond/Defrost)	XDX		Baseline		DMXV
	Test	Btu/day	Test	Btu/day	
-30 °F/75 °F/24 hour	XTC2	461,179	BSTC2	443,509	+4.0%
-30 °F/75 °F/48 hour	XTC3	434,829	BSTC3	453,257	-4.1%
-27 °F/75 °F/24 hour	XTC4	389,217	BSTC4	382,266	+1.8%
-30 °F/105 °F/24 hour	XTC5	506,682	BSTC5	489,063	+3.6%

TECHNICAL REPORT DATA

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15. SUPPLEMENTARY NOTES

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16. ABSTRACT

The report describes the performance, including energy consumption, of a refrigeration system incorporating a multifunctional valve (MXV) assembly. The MXV assembly (consisting of additional liquid line, an XTC valve, and a larger thermostatic expansion valve) was installed on all display cases of an instrumented supermarket refrigeration test rig. The refrigeration test rig includes two low- temperature single-deck display refrigerators; two 2-door reach-in cases; and a condensing unit with three unequal compressors, a water-cooled condenser, a water-cooled subcooler, an oil management system, and a programmable controller. Tests were performed at various combinations of evaporating temperature (-30 or -27 °F), condensing temperature (75 or 105 °F), and defrost schedules (once per 24 or 48 hours) under either temperature or pressure control. Results were compared to tests at the same conditions on the baseline system. Lower package temperatures were achieved under pressure control with the MVX assembly due to the lower superheats specified by the MXV manufacturer, but this reduction came at an energy penalty. Under temperature control—the control methodology used in field applications—there was no energy or product temperature benefit seen with the MXV valve assembly. Although coil pull-down times after defrost were shorter, there was no impact on daily energy use. Both the MXV and baseline system performed well with one defrost per 48 hours, and each had about 4% energy savings compared to a more frequent defrost schedule. However, at this condition, MXV showed no added benefit over baseline.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
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